ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

DESIGN AND NUMERICAL ENHANCEMENT ANALYSIS OF SLANTING TYPE BAFFLE PLATE IN SHELL AND TUBE HEAT EXCHANGER

V. Vedhagiri Eswaran[1], G. Ramadoss[2] and N. Subramanian[3]
[1]Quarters, Kongu Polytechnic College, Tamilnadu, India
[2]Shivani College of Engineering and Technology, Navallur Kuttapattu Village, Trichy, Tamil Nadu, India
[3]Department of Chemical Engineering, Kongu Poltechnic College, Perundirai, India
E-Mail: heattransfer.cfd@gmail.com

ABSTRACT

In this research work it consists Heat exchangers are important heat transfer apparatus in Food processing, Energy conversion, Thermal sectors etc. The present work modifies the existing bends flow and pressure Losses method used for conventional foam hybrid heat exchanger, apply a consideration the slant type geometry of slant-changer. Thermal analysis was carried out to study the impacts of various baffle slant angles on fluid flow and heat transfer of heat exchangers with Slant baffles. The analysis was conducted for conventional shell and tube heat Exchanger and Twist type heat changer for five baffle Slant angles. Numerical finite element approach Analysis results indicate that continual Slant Type baffles can reduce or even eliminate dead regions in the shell side of shell-and-tube heat exchangers. The pressure Losses varies drastically with baffle slant angle and shell-side Reynolds number. The variation of the velocity Losses is relatively large for small slant angle. However, for flat or straight type, 15°-55°, the effect of α on velocity Losses is very less. Different between to the segregate hybrid heat exchangers, the hybrid heat exchangers with continual Slant baffles have higher heat transfer coefficients to the same pressure ,velocity losses. The detailed Geometry on the heat convection and pressure Losses across the shell side will provide further basis flow for further optimization of shell-and-tube heat exchangers.

Keywords: slant baffles, bends flow pressure losses, twist angle, heat transfer coefficient per unit pressure Losses, shell and tube heat exchanger.

INTRODUCTION

Conventional heat exchangers with segmental baffles in shell side have some shortcomings resulting in the relatively low conversion of pressure Losses into a useful heat transfer.

Both hydrodynamic studies and testing of heat transfer and the pressure Losses on research facilities and industrial equipment showed much better performance of Slantly baffled heat exchanger when compared with conventional ones. These results in relatively high value of shell side heat transfer coefficient, low pressure Losses, and low shell side fouling [1].

Desirable features of heat exchangers

In order to obtain maximum heat exchanger performance at the lowest possible operating and capital costs without comprising the reliability, the following features are required of an Exchanger:

a) Higher heat transfer coefficient and larger heat transfer area.
b) Lower pressure Losses.

The objective of the present work is to determine the pressure Losses and heat transfer on shell side of Twist type heat changer analytically. A comprehensive experimental investigation on heat transfer and pressure Losses on Twist type heat exchanger is very expensive. The paper discusses developments in the Twist type heat changer and its design and research aspects. Important geometrical parameters have been discussed while calculating the thermal parameters. For calculating the pressure Losses and heat transfer on shell side of conventional as well as Slant baffle heat exchanger, Bell Delaware method with suitable modification has been used. Results and discussions shows that for all the Slant baffle heat exchangers studied, the ratios of heat transfer coefficient to pressure Losses are higher than those of a conventional segmental heat exchanger. This means that the heat exchangers with Slant baffles will have a higher heat transfer coefficient, when consuming the same pumping power.

Developments in shell and tube exchanger

The developments for shell and tube exchangers center on better conversion of pressure Losses into heat transfer by improving the conventional baffle designs. With single segmental baffles, a significant proportion of the overall pressure Losses is wasted in changing the direction of flow. This baffle arrangement also leads to other undesirable effects such as dead spots or zones of recirculation which can cause increased fouling, high leakage flow and large cross flow. The cross flow not only reduces the mean temperature difference but can also cause potentially damaging tube vibration [2].

Slant baffles heat exchanger or twist type heat exchanger

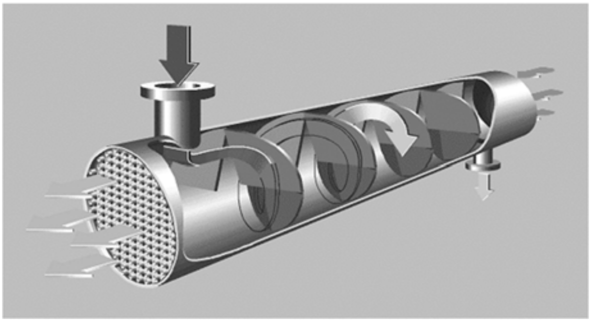

Figure-1. Slant baffle heat exchanger.

The baffles are of primary importance in improving mixing levels and consequently enhancing heat transfer of shell-and-tube heat exchangers. However, the segmental baffles have some adverse effects such as large back mixing, fouling, high leakage flow and large cross flow [3].

Compared to the conventional segmental baffled shell and tube exchanger Twist type heat changer offers the following general advantages [4].

a Increased heat transfer rate/ pressure Losses ratio.
b Reduced bypass effects.
c Reduced shell side fouling.
d Prevention of flow induced vibration.
e Reduced maintenance.

Problem identification

a) Therefore it is essential to develop a new type of shall tube heat exchanger using continuous type of baffle, which can have the following attributes [2]:
b) Improvement of shell side heat transfer.
c) Less pressure drop for a given mass flow rate.
d) Reducing of bypass effects in shell side.
e) Decreasing of fouling in shell side.
f) Prevention of bundle vibration.

Literature survey

Malcolm J. Andrews [1] in this research work its simulated with detailed three-dimensional computational fluid dynamics (CFD) simulations have been performed to explore the performance of a helically baffled heat exchanger, commercially referred to as the Helixchanger heat exchanger. The CFD simulations employ the HEATX computer simulation program, which is designed for the simulation of shell-and-tube heat exchangers.

Blaso[2] In this most causes the main barrier to flow of in shell and tube heat exchanger its convective resistance of inner and outer surface of the area increases effectiveness and number of assessments techniques wire coil insert, tubes flow twist, inserted and cylindrical tubes are considering to followed in experimentally.

A Dewan [3] Heat transfer augmentation techniques (passive, active or a combination of passive and active methods) are commonly used in areas such as process industries, heating and cooling in evaporators, thermal power plants, air-conditioning equipment, refrigerators, radiators for space vehicles, automobiles, etc. Passive techniques, where inserts are used in the flow passage to augment the heat transfer rate, are advantageous compared with active techniques, because the insert manufacturing process is simple and these techniques can be easily employed in an existing heat exchanger.

C.B. Sobhan *et al*. [4] experimentally investigated on a 1-2 shell and tube heat exchanger, to study the spiral turbulators on its performance. Experiments were conducted with various winding wire diameters and pitches and the heat transfer coefficient were evaluated for a wide range of temperature levels and flow rates of the shell side fluid. Nusselt number values for the various winding pitches were used. It is clearly understood that the use of the turbulators have considerable effect on the performance enhancement in the range the Reynolds numbers studied, which was in the laminar regime; improvement as high as 70% when compared to the bare tube has been noticed in the outside Nusselt number. The various pitches used, the best performance was observed for the case with 4 cm winding pitch, indicating the existence of an optimum winding pitch near this value. It was found that an increase in the wire diameter from 1.34 mm to 1.65 mm improves the overall Nusselt number to some extent, that is, the larger diameter wire effects slightly better heat transfer coefficient.

C. Yildiz *et al*. [5] were placed twisted narrow, thin metallic strips in the inner pipe of a concentric double-pipe heat exchanger and studied their effect on heat transfer and pressure drop for parallel and countercurrent flow. These turbulators were prepared by twisting the strips through certain angles and designed to touch the inside wall at each step. In the system, hot air was passed through the inner pipe, while cold water was flowing through the annulus. The experiments were performed with an empty inner pipe at Re number between 3400 and 6900. The effect of the turbulators on the heat transfer is more pronounced for high Re numbers. The improvements for parallel flow show a parallel trend and are only 10% lower than those for counter current flow. The increase is about 1.3 times that of the empty tube at the highest Re number for 170 mm pitch size.

Kenan Yakut *et al*. [6] investigated flow-induced vibration characteristics of conical-ring turbulators for heat transfer enhancement in heat exchangers experimentally. The conical- rings, having 10, 20 and 30 mm pitches, were inserted in a model pipe-line through which air was passed as the working fluid. It was observed that as the pitch increases, vortex shedding frequencies also increased and the maximum amplitudes of the vortices produced by conical-ring turbulators occur with small pitches. In addition, the effects of the promoters on the heat transfer and friction factor were investigated experimentally for all the arrangements. It was found that the Nusselt number increased with the increasing

ARPN Journal of Engineering and Applied Sciences

Reynolds number and the maximum heat transfer was obtained for the smallest pitch arrangement.

The pressure loss was much higher along a unit experiment element because there is an increase in the friction surfaces of these turbulators that also behave like a sequential loss element and work like a diffuser with respect to the arrangement positions.

V. Kongkaitpaiboon *et al*. [7] performed an experimental investigation of convective heat transfer and pressure loss in a round tube fitted with circular- ring turbulators. They studied the effect of the circular-ring turbulator (CRT) on the heat transfer and fluid friction characteristics in a heat exchanger tube. The experiments were conducted by insertion of CRTs with various geometries, including three different diameter ratios (DR=d/D=0.5, 0.6 and 0.7) and three different pitch ratios (PR=p/D=6, 8 and 12). During the test air at 27 °C was passed through the test tube which was controlled under uniform wall heat flux condition. The Reynolds number was varied from 4000 to 20, 000. According to the experimental results, heat transfer rates in the tube fitted with CRTs were augmented around 57% to 195% compared to that in the plain tube, depending upon operating conditions. Influence of the diameter ratio (DR) and pitch ratio (PR) on the heat transfer rate, friction factor and thermal performance factor behaviors was investigated under uniform wall heat flux condition. The CRTs with different diameter ratios (DR=d/D=0.5, 0.6 and 0.7) and pitch ratios (6, 8 and 12) were employed for the Reynolds number ranged between 4000 and 20,000. Over the entire range investigated CRTs propose heat transfer enhancement around 57% to 195% compared to that in the plain tube. The maximum thermal performance factor of 1.07 is found by the use of the CRT with DR=0.7 and PR=6.

V. Kongkaitpaiboon *et al*. [8] reported an experimental investigation of heat transfer and turbulent flow friction in a tube fitted with perforated conical-rings. They have been investigated the influences of the PCR on the turbulent convective heat transfer (Nu), friction factor (f) and thermal performance factor (η) characteristics experimentally. The perforated conical-rings (PCRs) used were of three different pitch ratios (PR=p/D=4, 6 and 12) and three different numbers of perforated holes (N=4, 6 and 8 holes). The experiment conducted in the range of Reynolds number between 4000 and 20, 000, under uniform wall heat flux condition and using air as the testing fluid. It was found that the PCR considerably diminishes the development of thermal boundary layer, leading to the heat transfer rate up to about 137% over that in the plain tube. Evidently, the PCRs can enhance heat transfer more efficient than the typical CR on the basis of thermal performance factor of around 0.92 at the same pumping power. Over the range investigated, the maximum thermal performance factor of around 0.92 was found at PR=4 and N=8 holes with Reynolds number of 4000. The effects of the pitch ratio (PR) and number of perforated hole (N) on the heat transfer enhancement in a tube are also considered. The concluding remarks can be described as follows:

(1) At the similar test conditions, the PCRs offers lower heat transfer enhancement than the CRs. However, they generate friction factor only around 25% of that produced by the PCRs.

(2) The heat transfer rate and friction factor of PCRs increase with decreasing pitch ratio (PR) and decreasing number of perforated hole (N).

(3) The mean heat transfer rates obtained from using the PCR with PR=4, 6, and 12 are found to be respectively, 185%, 140%, and 86%, over the plain tube. Over the range investigated, the maximum thermal performance factor of around 0.92 is found at PR=4 and N=8 holes with the Reynolds number of 4000.

Aydın Durmus [9] reported heat transfer and exergy loss in cut out conical turbulators. He investigated the effect of cut out conical turbulators, placed in a heat exchanger tube at constant outer surface temperature, on the heat transfer rates. The air was passed through the exchanger tube, the outer surface of which was heated with saturated water vapor. The experiments were conducted for air flow rates in the range of 15, 000 6 Re <60, 000. Heat transfer, pressure loss and exergy analyses were made for the conditions with and without turbulators and compared to each other. However, since the turbulators were placed directly in the flow area, they cause pressure losses. Therefore, an optimization should be made for pressure losses because pressure losses cause higher pumping power. Thus, future work may be undertaken on the detailed effect of the number of turbulators and on applications in flue and smoke tube boilers involving very hot gases.

Irfan Kurtbas *et al*. [10] devised a novel conical injector type swirl generator (CITSG). Performances of heat transfer and pressure drop in a pipe with the CITSG were experimentally examined for the CITSGs' angle (α) of 30°, 45° and 60° in Reynolds number (Re) range of 10, 000-35, 000. Moreover, circular holes with different numbers (N) and cross- section areas (Ah) were drilled on the CITSG. All experiments were conducted with air accordingly; Prandtl number was approximately fixed at 0.71. The local Nusselt numbers (Nux), heat transfer enhancement ratio (NuER) and heat transfer performance ratio (NuPR) were also calculated. It was found that the NuER decreases with increase in Reynolds number, the director angle (β), the director diameter (d), and with decrease in the CITSG angle (α). Likewise, variation of NuPR and NuER was also essentially similar for the same independent parameters.

P. Promvonge and S. Eiamsa-ard [11] have been investigated heat transfer, friction factor and enhancement efficiency characteristics in a circular tube fitted with conical-ring turbulators and a twisted- tape swirl generator experimentally. Air as the tested fluid was passed both enhancement devices in a Reynolds number range of 6000 to 26,000. Two twisted-tapes of different twist ratios, Y=3.75, and 7.5, were introduced in each run. The experimental results reveal that the tube fitted with the

ARPN Journal of Engineering and Applied Sciences

conical-ring and twisted-tape provides Nusselt number values of around 4 to 10% and enhancement efficiency of 4 to 8% higher than that with the conical-ring alone. A maximum heat transfer rate of 367% and enhancement efficiency of around 1.96 was found for using the conical-ring and the twisted-tape of Y=3.75. For all the devices used, the enhancement efficiency tends to decrease with the rise of Reynolds number and to be nearly uniform for Reynolds number over 16, 000. It was found that the smaller twist ratio was, the larger the heat transfer and friction factor for all Reynolds numbers. The average heat transfer rates from using both the conical-ring and twisted-tape for Y=3.75, and 7.5, respectively, were found to be 367% and 350% over the plain tube. However, the friction factor from using both devices also increases considerably. S. Eiamsa-ard *et al*. [12] have been investigated heat transfer, flow friction and thermal performance factor characteristics in a tube fitted with deltawinglet twisted tape, using water as working fluid experimentally. Influences of the oblique delta- winglet twisted tape (O-DWT) and straight delta- winglet twisted tape (S-DWT) arrangements were also described. The experiments were conducted using the tapes with three twist ratios (y/w = 3, 4 and 5) and three depth of wing cut ratios (DR = d/w = 0.11, 0.21 and 0.32) over a Reynolds number range of 3000-27, 000 in a uniform wall heat flux tube. The obtained results show that mean Nusselt number and mean friction factor in the tube with the delta-winglet twisted tape increase with decreasing twisted ratio (y/w) and increasing depth of wing cut ratio (DR). It was also observed that the O-DWT was more effective turbulator giving higher heat transfer coefficient than the S-DWT. Over the range considered, Nusselt number, friction factor and thermal performance factor in a tube with the O-DWT were, respectively, 1.04-1.64, 1.09-1.95, and 1.05-1.13 times of those in the tube with typical twisted tape (TT). Empirical correlations for predicting Nusselt number and friction factor have been employed. The predicted data were within ±10% for Nusselt number and ±10% for friction factor.

A. Durmus *et al*. [13] investigated the effect of propeller type turbulators which were located in the laminar pipe of co axial heat exchanger. The blade angle of propeller of turbulator between 10≤θ≤40 the ration of propeller diameter to pipe diameter between 0.87≤Dk /Db≤0.94 and Reynolds number in the range of 10000and 30000.The turbulator place with 10 cm array increased the heat transfer as much as 28% and 39% the pressure loss as much as 17% and 43% compared with the turbulator placed in the arrays of 20 and 30cm. The turbulator with blade angle of 200 decrease heat transfer as far as 8%, the ones with a blade angle of 400, on the other hand, as far as 35% compared with the turbulators with a blade angle of 100. The change of blade angle affected the pressure loss between 15-40%. In the experiment was seen that heat transfer 2-4 times and pressure loses 8.5 times biggest than the values of the empty pipe heat exchanger at Reynolds number of 10000-30000 and different mass flow rate.

Anil Singh Yadav [14] has been studied the influences of the half length twisted tape insertion on heat transfer and pressure drop characteristics in a U- bend double pipe heat exchanger experimentally. In the experiments, the swirling flow was introduced by using half-length twisted tape placed inside the inner test tube of the heat exchanger. The experimental results revealed that the increase in heat transfer rate of the twisted-tape inserts was found to be strongly influenced by tape-induced swirl or vortex motion. The heat transfer coefficient was found to increase by 40% with half-length twisted tape inserts when compared with plain heat exchanger. It was also observed that the thermal performance of Plain heat exchanger was better than half length twisted tape by 1.3-15 times.

Piroz Zamankhan [15] has been developed a 3D mathematical model to investigate the heat transfer augmentation in a circular tube with a helical turbulator. Glycol-water blends of various concentrations were used in the inner tube, and pure water was used in the outer tube. Changes in fluid physical properties with temperature were taken into account, and k-ε; k-ω, and large eddy simulations were developed for turbulence modeling. The simulation results showed a nonlinear variation in Reynolds and Prandtl numbers for a long model of a heat exchanger even in the absence of a turbulator. The presence of the turbulator was found to increase the heat transfer, sometimes without inducing turbulence, but also increased the pressure drop. Comparing their numerical results with experimental results, it was found that the LES model predicts the behavior of real systems. Using a multi-objective optimization method such as Genetic Algorithm coupled with the GPU-LES-SPH (which is an implementation of Smoothed Particle Hydrodynamics), a set of solutions will be obtained that satisfy different levels of compromise. From this set the most suitable solution will be selected. This efficiency-enhancing tool developed will be particularly suitable for process intensification (i.e., increasing production capacity per unit volume of equipment) in the chemical process industry.

Pongjet Promvonge [16] has been presented the experimental results on heat and flow friction characteristics in a uniform heat flux tube equipped with the 5 mm wire coil of three different coil pitch ratios (CR = 4, 6 and 8) and the twisted tapes of 2 twist ratios (Y = 4 and 6) are presented in the form of Nusselt number and friction factor. The Nusselt numbers obtained under turbulent flow conditions for the twisted tape and coiled wire with the three ratios.

Smith Eiamsa-ard and Pongjet Promvonge [17] have been conducted the experiments to investigate the heat transfer and friction factor characteristics of the fully developed turbulent airflow through a uniform heat flux tube fitted with diamond-shaped turbulators in tandem arrangements. Nusselt numbers along the tube fitted with the D-shape turbulator for cone angles θ=450 and tail length ratio TR = 1.0, for Reynolds numbers ranging from 3568 to 16, 228. Nusselt number was high at the entry region (7 ≤ X/D ≤ 22) and gradually decreased but at exit region (X/D > 22) slightly increases. Heat transfer variation in terms of Nusselt number of the inserted tube for different the included cone angle (θ=150, 300 and 450) with Reynolds numbers at TR = 1.0, 1.5 and 2.0, respectively. The larger angle (θ) yields a higher heat transfer rate than the smaller angle. The use of smaller the cone angle of θ =150 and 300 leads to the decrease in Nusselt number at 16.3% and 7.4%, respectively, in comparison with θ = 450 of all range Reynolds numbers studied. The friction factor variations of using the three included cone angles (θ = 150, 300 and 450) with Reynolds number between 3500 and 16,500 for TR = 1.0, 1.5 and 2.0, respectively. The maximum increase in friction factor was seen at TR = 1.0 which was higher than TR =1.5 and 2.0 around 11.3% and 22.6%, respectively. The improvement of average heat transfer rate was respectively 32.1%, 46.5% and 57.8% higher than those the plain tube while the friction factor was 4.7, 5.25 and 5.67 times of the plain tube.

Panida Seemawute and Smith Eiamsa-Ard [18] have been conducted the experiments for heat transfer in heat exchanger tubes by means of TRs compared to that of CRs at different width and pitch ratio has been investigated for Reynolds number between 6000 and 20, 000. At the same width ratio (W/D=0.15) and a given pitch ratio, only TRs with the smallest pitch ratio (p/D) of 1.0 give higher Nusselt numbers than the CRs by around 3 to 4%. TRs and CRs at p/D=1.5 yield comparable Nusselt numbers for the whole range tested. At p/D=2.0, Nusselt numbers attributed to the TRs become lower than those associated by the CRs. The Nusselt number associated by the TR at the largest width ratio (W/D=0.15), are augmented by around 35.7% and 60% over those associated by ones with W/D=0.10 and 0.05, respectively. By the utilization of TR at the smallest pitch ratio (p/D) of 1.0, an average Nusselt number was found to be around 6.8%and 13.6% higher than those achieved by the use of the ones of larger pitch ratios of 1.5 and 2.0 respectively. Thermal performance factor results Among the TRs tested, it was found that a thermal performance factor increased with decreasing width ratio. The TRs with W/D=0.05 at p/D=1.0, 1.5 and 2.0 give thermal performance factor in the ranges of 1.02 to 1.24, 1.01 to 1.2, and 1.00 to 1.2 respectively.

Panida Seemawute *et al*. [19] has been investigated comparatively visualization of flow characteristics induced by twisted tape consisting of alternate-axis (TA) to that induced by typical twisted tape (TT). The visualization was carried out via a dye injection technique. The TAs were made of aluminum strips (for heat transfer setup) and acrylic sheet (for visualization setup) with thickness of 1.0 mm (δ), width of 18 mm and length of 1000 mm. Straight tapes were prepared at three desired twist lengths in 180° rotation (y/W = 3, 4 and 5) by twisting straight tapes, about their longitudinal axis, while being held under tension. The heating test tube was made of copper with thickness of 1.5 mm, inner diameter of 19 mm and length of 1000 mm. A common swirl flow was induced by TTs. with TAs, fluid stream which directly encounters a crosswise edge of the tape at an alternate point, was separated. Effect of TAs at various twist ratios, y/W = 3, 4 and 5 on the Nusselt number. This result was corresponded to the superior chaotic mixing as described above, and consequently results in more violent interruption on thermal boundary layer, and thus more efficient heat transfer though the tube wall.

P. K. Nagarajan *et al*. [20] have been presented experimental investigation of heat transfer and friction factor characteristics of solar trough collector fitted with full length twisted tapes inserts of twist ratio 6, 8 and 10. The transitional flow regime was selected for this study with the Reynolds number range 1192 to 2534. Friction factor decreased with increase in Reynolds number for a given twist ratio. At Reynolds number 2000 we can able to see the transition where there is a steady friction factor value. The swirl flow caused by the twisted tape effectively increases the heat transfer. The Nusselt Number was increased with decreasing twist ratio. The experimental data obtained were compared with those obtained from plain tube published data. The use of twisted inserts improved the performance of solar collector. The cost involved for manufacturing and inserting twisted tape was very minimal compared to energy efficiency improvement provided by these inserts.

S. Naga Sarada *et al*. [21] presented the study an experimental investigation of the potential of reduced width twisted tape inserts to enhance the rate of heat transfer in a horizontal circular tube with inside diameter 27.5mm with air as working fluid. The twisted tapes were of three different twist ratios (3, 4 and 5) each with five different widths (26-full width, 22, 18, 14 and 10 mm) respectively. The Reynolds number varied from 6000 to 13500. The percentage increase in Nusselt numbers for reduced width tapes compared to plain tube were about 11-22%, 16-31%, 24-34% and 39-44% respectively for tape widths of 10, 14, 18 and 22 mm respectively for twist ratio =3.For full width tapes, the percentage increase was observed to be 58 to 70% compared to plain tube. The percentage increase in Nusselt numbers for reduced width tapes compared to plain tube are about 5-12%, 9-22%, 13-30% and 23-6% respectively for tape widths of 10, 14, 18 and 22 mm respectively for twist ratio =4. For full width tapes, the percentage increase was observed to be 36 to 42% compared to plain tube. The percentage increase in Nusselt numbers for reduced width tapes compared to plain tube were about 2-8%, 6-12%, 9-19% and 14-27% respectively for tape widths of 10, 14, 18 and 22 mm respectively for twist ratio =5. For full width tapes, the percentage increase was observed to be 22 to 37% compared to plain tube. The overall enhancement ratio of the tubes with full width twisted tape inserts was 1.62 for full width–26mm and 1.39 for reduced width-22mm twisted tape insert.

S.K. Saha *et al*. [22] have been investigated heat transfer and pressure drop characteristic in a circular tube fitted with regularly spaced twisted tape elements experimentally. For small cases the friction factor is 5-

VOL. 10, NO. 8, MAY 2015 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

10% less and heat transfer by 20-40%. The local Nusselt number remains almost constant in both the tape and annular sections. For the small diameter negative axial velocity were not predicted even at first axial step in the tape section and thus the swirl was not supported. The friction factor with the Reynolds number for varied y and s and for each case. The nusselt number decreases for w = 0.727 compared to that for w = 0.909 for all twist ratio and space ratio. For y = 5 and s = 5, the reduction was 25-32%; for y = 5 and s = 2.5, the reduction was 25-45%; for y = 2.5 and s = 5, the reduction was 22-45% and for y = 2.5 and s = 2.5, the reduction was 13-43%. The effect of the phase- angle between two successive tape elements on the friction factor and nusselt number increases were 8% and 15% respectively. The variation isothermal and heated friction factor with Reynolds number were 7% and 6%.

S. Eiamsa-ard and P. Promvonge [23] have been conducted an experimental study to investigate heat transfer and friction loss behaviors in a circular wavy surfaced tube with a helical-tape insert using hot air as the test fluid. In general, the average heat transfer rate for the wavy-surfaced tube with the helical tape was found to be 23 to 35% better than that for the wavy-surfaced tube alone. The corresponding increase in the mean Nusselt number of the wavy- surfaced tube with the helical tape was about 330% to 422% over the plain tube. For the wavy surfaced tube with the helical tape, the increase in friction factor was found to be around 50% above one without the tape the average heat transfer rate for employing the tape with rod was found to be 8 to 12% better than that for one without core-rod. Thus, for the tape without rod, the friction factor could be reduced around 50% below one with core-rod. Results of the present correlations reasonably agree well within ±10% in comparison with experimental data for the friction factor, and within ±10% for the Nusselt number. The maximum increase in heat transfer rate and friction factor were found to be about 4.2 and 110 times the plain tube for the flow range studied.

M.R. Salimpour and S. Yarmohammadi [24] have been conducted an experimental investigation to find the influence of twisted tape inserts on the pressure drop during forced convective condensation of R-404A vapor in a horizontal tube. The tube set 5 with twist ratio of 4 has the highest pressure drop. Reduction in twist ratio induces higher turbulence intensity in liquid film and vapor core. The condensing pressure drop for tube set 5 is up to 239% more than that for plain tube at refrigerant mass velocity 106.8 $kgms^{-2}$. Tube set 2 has the lowest range of pressure drop increment which increases pressure drop up to 151% compared to the plain tube for refrigerant mass velocity 89 $kgms^{-2}$. This correlation predicts experimental data with an error range of ±20%.

Shashank S. Choudhari and S.G. Taji [25] have been studied the experimental investigation of the heat transfer and friction factor characteristics of a double pipe heat exchanger fitted with coil wire insert made up of three different material as copper, aluminum and stainless steel and different pitches for Reynolds number in range of 4000-13000. Cu tube cause higher heat transfer enhancement about 1.58, and aluminum and stainless steel causes heat transfer rate enhancement up to 1.41 and 1.31 respectively. Overall heat transfer coefficient was higher for copper coil wire insert than aluminum, stainless steel inserts and plain tube. The friction factor of aluminum coil wire insert of 5 mm pitch was 5.4 to 6.7 times of the plane tube. Stainless steel tube insert causes friction factor of 4.8 to 5.9 times to plane tube and copper insert has friction factor of 4.3 to 5.4 times plane tube.

S. Selvam *et al*. [26] carried on experiments for different twist pitch to the width of the twisted tape ratios (y/w) for TTP and TTPB. The variation of Nusselt number with Reynolds number for the tube fitted with TTP of three different y/w ratios (3.33, 4.29, and 5.71). The experimental results it was seen that the smaller y/w (3.33) yields the higher values of heat transfer of about 23.86% than plain tube. Similarly for y/w=4.29 and 5.71 the enhancement were 19.9% and 14.4% respectively. It can be seen that the friction factor for y/w=4.29 and 5.71 were less when compared with y/w=3.33. This is due to less contact surface area of the turbulator. The empirical correlations developed relating pitch and Reynolds number were matching with the experimental data within ±7.28, and ±7.16% for Nusselt number and friction factor respectively.

Research background

Research on the Twist type heat changer has forced on two principle areas.

a) Hydrodynamic studies on the shell and side.
b) Heat transfer and pressure Losses studies on small scale and full industrial scale equipment.

RESEARCH METHODOLOGY

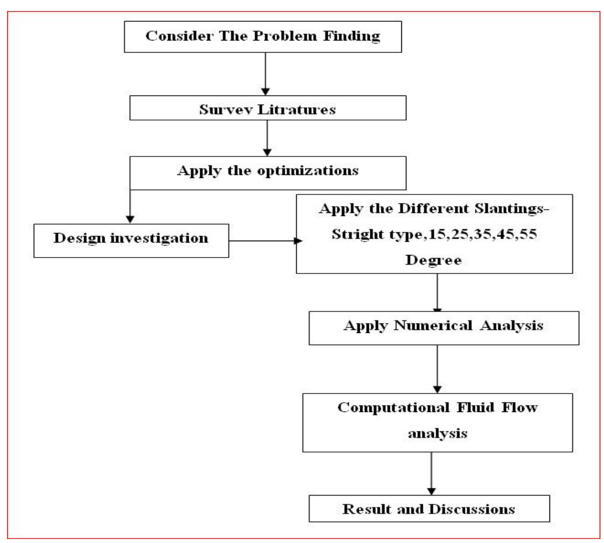

Figure-2. Research methodology process.

Design consideration

ARPN Journal of Engineering and Applied Sciences

An optimally designed Slant baffle arrangement depends largely on the heat exchanger operating conditions and can be accomplished by appropriate design of Twist type heat angle, baffle overlapping and tube layout [4-5].

In the original method for conventional shell and tube Heat exchanger, an ideal shell-side heat transfer coefficient is multiplied by various correction factors for flow distribution and the non-idealities such as leakage streams, bypass stream etc. are taken into consideration.

For Slant baffle geometry it is suggested that some correction factor are not required and suitable modification is done in the Bell Delaware method [4].

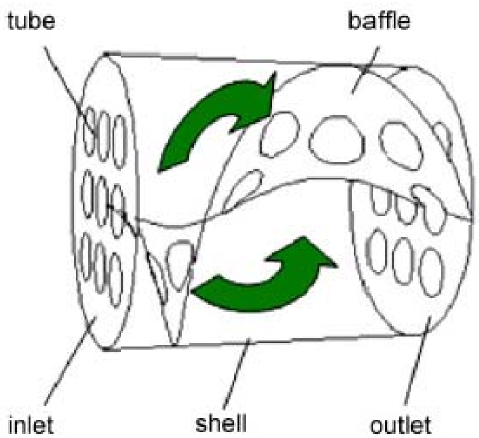

Figure-3. Twist type heat changer pitch.

Parameters Taken:

- Pressure Losses (PS)
- Baffles pitch (Twist type heat Transfer)
- Baffles angle (α)
- Baffle space (B)
- Surface area (A)
- Heat transfer coefficient (ho)

In designing of Twist type heat changer, pitch angle, baffle arrangement, and the space between two baffles with the same position are important parameters. Baffle pitch angle (α) is the angle between flow and perpendicular surface on exchanger axis and B is space between two following baffles with the same situation.

Changing the pitch angle in Slant baffle system can create wide range of flow velocities as compared to the baffle space and baffle cut in traditional heat exchangers. Moreover, overlapping of Slant baffles is a parameter that can affect significantly on shell side flow pattern.

A comprehensive study on heat transfer coefficient and pressure Losses characteristics of Segmental baffle Shell and Tube heat exchanger and Twist type heat changer has been done. The performance of Twist type heat changer for different Twist type heat angles are calculated and compared with Segmental heat exchanger and result is tabulated.

Analytical form heat transfer calculation-heat transfer

Heat transfer coefficients and pressure Losses calculations are the main part of design of heat exchangers with a given duty. In traditional approaches such as Kern and Bell–Delaware methods are used to calculate the heat transfer coefficient and pressure Losses.

For conventional tubular Heat Exchangers, Kern method, which was an attempt to correlate data for standard exchangers by a simple equation analogous to equations for flow in tubes. However, this method is restricted to a fixed baffle cut (25%) and cannot adequately account for baffle-to-shell leakages. Nevertheless, although the Kern equation is not particularly accurate, it does allow a simple and rapid calculation of shell side coefficients and pressure Losses to be carried out and has been successfully used since its inception.

The next stage of development of shell side calculation methods was that commonly described as Bell-Delaware method (Bell-1963). In this method, correction factors for baffle leakage effects, etc., are introduced based on experimental data. This method is widely used and is the basis of the approach recommended, in the Heat exchanger design handbook [5-7].

Thermal analyses of segmental baffle heat exchanger and twist type heat changer

In the present work, Bell Delaware method has been used for comparing the thermal performance of Segmental baffle heat exchanger and Twist type heat changer. Firstly, thermal parameters have been calculated for Segmental baffle heat exchanger using various steps mentioned by Bell-Delaware. Then, suitable modifications have been carried out in different steps for calculating the thermal parameters of Twist type heat changer. Then comparative analysis has been done between the two heat exchangers. Following data has been assumed and the geometrical parameters are kept constant for both types of heat exchangers.

Figure-4. Shall and tube heat exchanger.

Table-1. Data instalization-shell side.

S. No.	Parameter taken	Dimensions

VOL. 10, NO. 8, MAY 2015 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

1	Fluid	Water
2	Volume flow rate (Qv)	60 lpm
3	Mass flow rate (m)	1Kg/s
4	Shell ID (D_s)	0.153 m
5	Shell length	1.123m
6	Tube pitch	0.0225 m
7	No. of passes	1
8	Baffle cut	25% =0.25
9	Baffle pitch	0.060 m
10	Nozzle ID	0.023 m
11	No. of baffles	5
12	Mean Bulk Temperature	30°C
13	Tube OD	
14	Tube thickness	
15	No of tube	
16	Twist type heat angle	25^0
17	No of baffles for Twist type heat changer	5

Table-2. Data instalization-tube side.

S. No.	Parameter taken	Dimensions
1	Fluid	Water
2	Volume flow rate (Qv)	10 lpm
3	Mass flow rate (m)	0.17 Kg/s
4	Nozzle ID	0.023 m
5	Mean Bulk Temperature	C
6	Tube OD	0.012 m
7	Tube thickness	0.0014 m
8	No. of tube	24

Instalization fluid properties

Table-3. Fluid properties.

Properties	**Cold water** (Shell side)	**Hot water** (Tube side)
Heat capacity	4.178 kJ/Kg k	4.178 kJ/Kg k
Thermal conductivity	0.6150 W/m. K	0.6150 W/m. K
Fluid viscosity	0.001 Kg.s/m^2	0.001 Kg.s/m2
Sp Gr.	0.996	0.996
Prantle No.	5.42	5.42
Density	996 kg/m^3	996 kg/m^3

Leakage and pass clearances

Table-4. Leakage pass clearance.

S. No.	Parameter taken	Dimensions [M]
1	Tube to baffle clearance	0.004
2	Baffle to shell clearance	0.001
3	Shell To bundle clearance	0.01428
4	Shell outer tube Limit	0.1387
5	Shell centre tube limit	0.1267

RESULT AND DESCUSSIONS

It shows that, from Table-3, the pressure Losses for Segmental baffle heat exchanger is 2.209 KPa and for Twist type heat changer (15^0) it is 0.645 KPa and it is minimum for all Twist type heat angles.

Also the, ratio of heat transfer coefficient to pressure Losses for Twist type heat changer (25^0) is 3.90 and for Segmental baffle heat exchanger it is Tables 1-7. This ratio for Twist type heat changer is about three times higher than Segmental baffle heat exchanger graphs [5-12] as shown in Figure.

Table-5. Twist type Vs shell side renolds number.

S. No.	Twist type of heat exchanger	Cross flow area (m^2)
01	0° [straight type]	0.004407
02	15°	0.004721
03	25°	0.008224
04	35°	0.012373
05	45°	0.480667
06	55°	0.686437

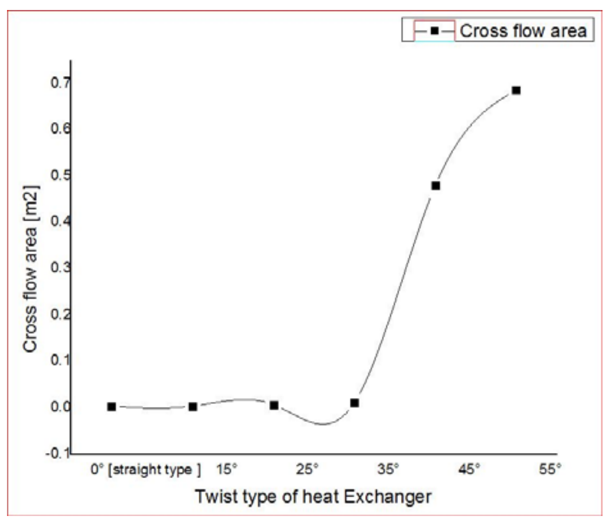

Figure-5. Twist type Vs shell side renolds.

ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

Table-6. Twist type Vs shell side renolds number.

S. No.	Twist type of heat exchanger	Shell side renolds number
01	0° [straight type]	2727
02	15°	2738
03	25°	1471
04	35°	172
05	45°	27
06	55°	1

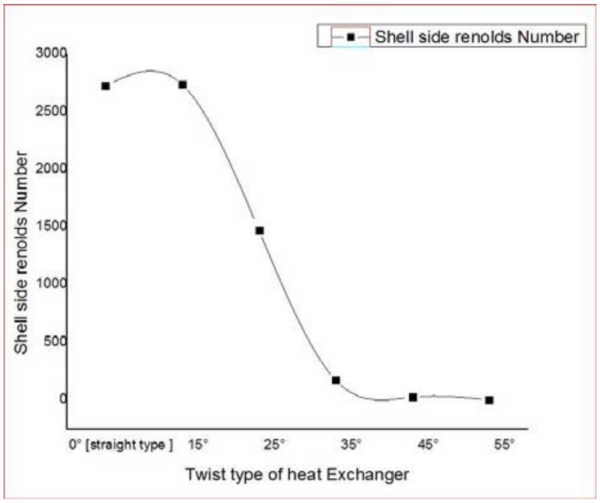

Figure-6. Twist type Vs shell side renolds number.

Table-7. Twist type Vs Shell side heat transfer of Coefficient (W/m^2.k).

S. No.	Twist type of heat exchanger	Shell side heat transfer of Coefficient (W/m^2.k)
01	0° [straight type]	2770.82
02	15°	2487.317
03	25°	1631.67
04	35°	1331.371
05	45°	122.7814
06	55°	71.677

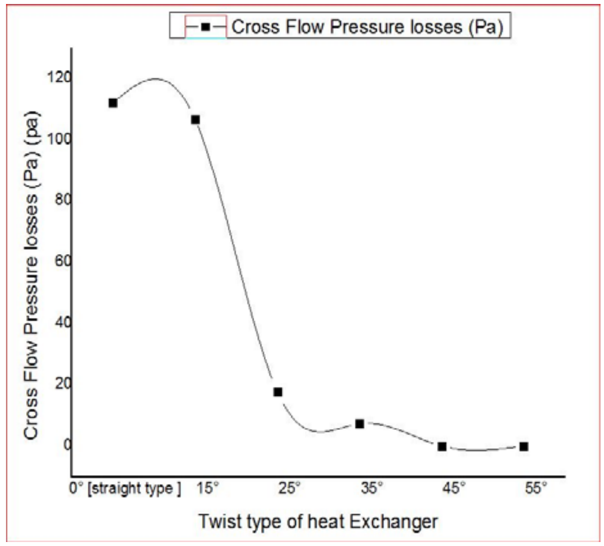

Figure-7. Twist type Vs shell side heat transfer of Coefficient (W/m^2.k).

Table-8. Twist type Vs Cross Flow Pressure losses (Pa).

S. No	Twist type of heat exchanger	Cross flow pressure losses (Pa)
01	0° [straight type]	112.22
02	15°	106.7728
03	25°	17.83
04	35°	7.4248
05	45°	0.064337
06	55°	0.074337

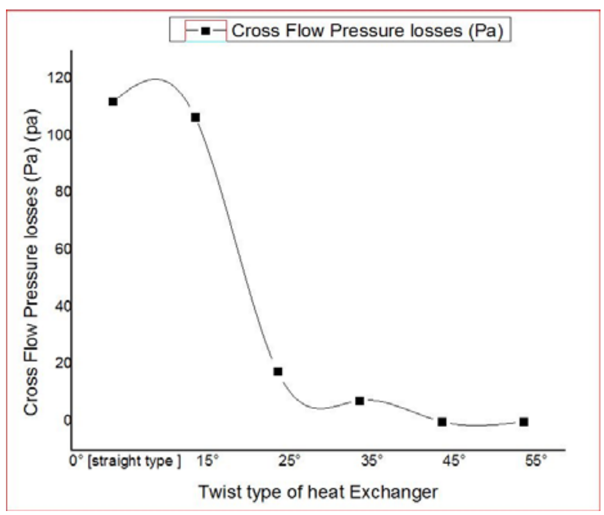

Figure-8. Twist type Vs cross flow pressure losses (Pa).

ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

Table-9. Twist type Vs Pressure losses on baffle window (Pa).

S. No.	Twist type of heat exchanger	Pressure losses on baffle window (Pa)
01	0° [straight type]	102.24
02	15°	0.2181
03	25°	0.1711
04	35°	0.1144
05	45°	0.002141
06	55°	0.002341

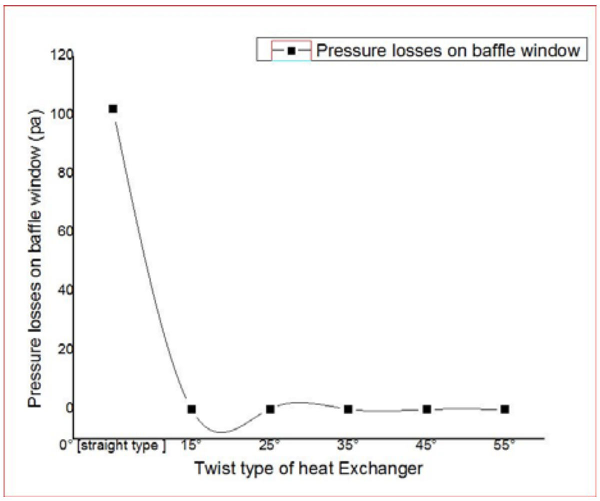

Figure-9. Twist type Vs pressure losses on baffle window(Pa)

Table-10. Twist type Vs Total shell side pressure losses.

S. No.	Twist type of heat exchanger	Total shell side pressure losses
01	0° [straight type]	2201.6
02	15°	674.7
03	25°	420.7
04	35°	24.21
05	45°	0.2134
06	55°	0.2734

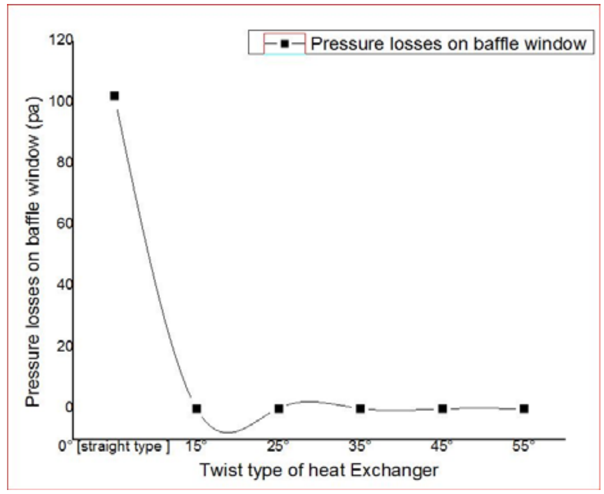

Figure-10. Twist type Vs total shell side pressure losses.

CONCLUSIONS

In this paper, the thermal analysis for heat exchangers with different baffle slant angles are performed to show the effects of baffle slant angle on the heat transfer and pressure Losses characteristics. The major findings are summarized as follows:

a. The flow pattern in the shell side with continual Slant baffle is near-plug flow. Therefore, the dead region is eliminated and the heat transfer area is used more effectively.
b. As the shell-side Reynolds number is increased, the pressure Losses increases for all the cases considered. For all Slant baffle heat exchangers studied, the pressures Lossess are lower than those of the conventional segmental heat exchangers. The pressure Losses decreases with the increase of baffle slant angle in all the cases considered. The change of the pressure Losses is large in the small slant angle region. However, the effects of baffle slant angle on pressure Losses are small when α>350.
c. For all the Slant baffle heat exchangers studied, the ratios of heat transfer coefficient to pressure Losses are higher than those of a conventional segmental heat exchanger. This means that the heat exchangers with Slant baffles will have a higher heat transfer coefficient, when consuming the same pumping power.
d. It can be concluded that proper baffle slant angle will provide an optimal performance of heat exchangers. The detailed knowledge of the heat transfer and flow distribution provided in this investigation may serve as a basis for further optimization of shell-and-tube heat exchangers
e. Bell Delaware method available in the literature is only for the segmental baffle heat exchanger, since Slant baffle heat exchanger is the recent development. Suitable modifications in Bell Delaware method can

give the results for Twist type heat changer and can serve as a preliminary method for thermal analysis.

Future scope

In the future this report could be extended by analyzing the performance of inertial navigation systems under Thermal motion, rather than just looking at the stationary case slanting type heat exchanger. In particular quantization errors become much more significant more vibrations mass flow rate and etc., when the device is undergoing rotations baffles choice of integration scheme can have an big effect on the performance of the system. Other possible future work includes extending the simulator described in this report to model more sources of error finding numerical aspects to apply the experimental mode.

REFERENCES

[1] Andrews Malcolm, Master Bashir. 2005. Three Dimensional Modeling of a Twist type heat changer Heat exchanger using CFD. Heat Transfer Engineering. 26(6): 22-31.

[2] C.B. Sobhan, K.T. Mohammed kutty, M. Hannan and P. Krishtaiah. 1994. Experimental investigations on a 1-2 heat exchanger with wire-wound tubes. warme- und Stoffubertragung. 29: 211-217.

[3] Cengiz Yildiz, Yasar Bicer and Dursun Pehlivan. 1998. Effect of Twisted Stripes on Heat Transfer and Pressure Drop in the Heat Exchangers. Energy Conversion and Management. 39: 331-336.

[4] Kenan Yakut and Bayram Sahin. 2004. Flow-induced vibration analysis of conical rings used for heat transfer enhancement in heat exchangers. Applied Energy. 78: 273-288, doi:10.1016/j.apenergy. 2003.09.001.

[5] V. Kongkaitpaiboon, K. Nanan and S. Eiamsa-ard. 2010. Experimental investigation of convective heat transfer and pressure loss in a round tube fitted with circular-ring turbulators. International Communications in Heat and Mass Transfer. 37: 568-574, doi:10.1016/j.icheatmasstransfer.2009.12.016.

[6] V. Kongkaitpaiboon, K. Nanan and S. Eiamsa-ard. 2010. Experimental investigation of heat transfer and turbulent flow friction in a tube fitted with perforated conical-rings. International Communications in Heat and Mass Transfer. 37: 560-567, doi:10.1016/j.icheatmasstransfer.2009.12.015.

[7] Aydın Durmus. 2004. Heat transfer and exergy loss in cut out conical turbulators. Energy Conversion and Management. 45: 785-796, doi: 10.1016/S0196-8904(03)00186-9.

[8] I. Kurtbas, F. Gülçimen, A. Akbulut and D. Buran. 2009. Heat transfer augmentation by swirl generators inserted into a tube with constant heat flux. International Communications in Heat and Mass Transfer. 36: 865-871, doi:10.1016/j.icheatmasstransfer. 2009.04.011.

[9] P. Promvonge and S. Eiamsa-ard. 2007. Heat transfer behaviors in a tube with combined conical-ring and twisted-tape insert. International Communications in Heat and Mass Transfer. 34: 849-859, doi:10.1016/j.icheatmasstransfer.2007.03.019.

[10] S. Eiamsa-ard, K. Wongcharee, P. Eiamsa-ard and C. Thianpong. 2010. Heat transfer enhancement in a tube using delta-winglet twisted tape inserts. Applied Thermal Engineering. 30: 310-318, doi:10.1016/j.applthermaleng.2009.09.006.

[11] A. Durmus, I. Kurbas, F. Gulcimen and E. Turgut. 2004. Investigation of the effect of co-axis free rotating propeller-type turbulators on the performance of heat exchanger. International Communications in Heat and Mass Transfer. 31: 133-142.

[12] Anil Singh Yadav. 2009. Effect of Half Length Twisted-Tape Turbulators on Heat Transfer and Pressure Drop Characteristics inside a Double Pipe U-Bend Heat Exchanger. Jordan Journal of Mechanical and Industrial Engineering. 3: 17- 22.

[13] Piroz Zamankhan. 2010. Heat transfer in counterflow heat exchangers with helical turbulators. Commun Nonlinear Sci Numer Simulat. 15: 2894-2907, doi:10.1016/j.cnsns.2009.10.025.

[14] P. Promvonge. 2008. Thermal augmentation in circular tube with twisted tape and wire coil turbulators. Energy Conversion and Management. 49: 2949-2955, doi:10.1016/j.enconman.2008.06.022.

[15] S. Eiamsa-ard and P. Promvonge. 2010. Thermal characterization of turbulent tube flows over diamond-shaped elements in tandem. International Journal of Thermal Sciences. 49: 1051-1062, doi:10.1016/j.ijthermalsci.2009.12.003.

[16] C. Thianpong, K. Yongsiri, K. Nanan and S. Eiamsa-ard. 2012. Thermal performance evaluation of heat exchangers fitted with twisted-ring turbulators. International Communications in Heat and Mass Transfer. 39: 861-868. Doi: 10.1016/ j.icheatmasstransfer.2012. 04.004.

[17] P. Seemawute and S. Eiamsa-Ard. 2012. Visualization of Flow and Heat Transfer in Tube with Twisted Tape Consisting of Alternate Axis. 4th International Conference on Computer Modeling and Simulation. 22: 36-40.

[18] P.K. Nagarajan, Nitesh Mittal and Rohit Chechani. 2010. Experimental studies on heat transfer and friction factor characteristics of parabolic trough solar water heating system with and without twisted tapes. Proceedings of the 37th National and 4th International Conference on Fluid Mechanics and Fluid Power. December. pp. 16-18.

[19] S. Naga Sarada, A. V. Sita Rama Raju, K. Kalyani Radha and L. Shyam Sunder. 2010. Enhancement of heat transfer using varying width twisted tape inserts. International Journal of Engineering, Science and Technology. 2: 107-118.

[20] S. K. Saha, A. Dutta and S. K. Dhal. 2011. Friction and heat transfer characteristics of laminar swirl flow through a circular tube fitted with regularly spaced twisted-tape elements. International journal of Heat and Mass Transfer. 44: 4211-4223.

[21] S. Eiamsa-ard and P Promvonge. 2007. Enhancement of Heat Transfer in a Circular Wavy-surfaced Tube with a Helical- tape Insert. International Energy Journal. 8: 29-36.

[22] M. R. Salimpour and S.m Yarmohammadi. 2012. Effect of twisted tape inserts on pressure drop during R-404A condensation. International journal of refrigeration. 35: 263-269.

[23] S. Choudhari Shashank and S.G. Taji. 2013. Experimental Studies on Effect of Coil Wire Insert on Heat Transfer Enhancement and Friction Factor of Double Pipe Heat Exchanger. International Journal of Computational Engineering Research. 3: 32-39.

[24] S. Selvam, Pr. Thiyagarajan and S. Suresh. 2013. Experimental Studies on Effect of Bonding the Twisted Tape with Pins to the Inner Surface of the Circular Tube. Thermal Science, 2013, OnLine-First (00):36-36. DOI: 10.2298/TSCI120807036S.

[25] Panida Seemawute, and Smith Eiamsa-ard. 2010. Thermohydraulics of turbulent flow through a round tube by a peripherally-cut twisted tape with an alternate axis. International Communications in Heat and Mass Transfer. 37: 652-659, DOI: doi:10.1016/j.icheatmasstransfer.2010.03.005.